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DESCRIPTION

RANKINE CYCLE SYSTEM

FIELD OF THE INVENTION

The present invention relates to a Rankine cycle system that includes an evaporator for heating a liquid-phase working medium with exhaust gas of an engine so as to generate a gas-phase working medium, and a displacement type expander for converting the thermal energy of the gas-phase working medium generated in the evaporator into mechanical energy.

BACKGROUND ART

Japanese Patent Application Laid-open No. 2000-345835 discloses a waste heat recovery system for driving a turbine by heating coolant vapor of a cooling system of an engine with waste heat of the engine, in which the thermal efficiency is enhanced by optimally controlling the pressure or temperature of a cooling path according to engine running conditions. Specifically, the value for a target pressure of the cooling path is lowered as the engine rotational speed and the engine load increase, and the amount discharged from a coolant circulation pump, etc. is controlled so that the actual pressure coincides with the target pressure.

In a Rankine cycle system equipped with a displacement type expander, as shown in FIG. 4, if the steam pressure at the inlet of the expander coincides with a target steam pressure (optimum steam pressure), the steam pressure at the outlet of the expander becomes a pressure that is commensurate with the expansion ratio of the expander, but if the steam pressure at the inlet is too high, there is the problem that the steam discharged from the outlet of the expander has surplus energy remaining and the energy is wastefully discarded. On the other hand, if the steam pressure at the inlet is

too low, there is the problem that the pressure of the steam discharged from the outlet of the expander becomes negative and the expander carries out negative work, thus degrading the efficiency.

Although it is important to make the steam pressure supplied to the expander coincide with a target steam pressure in this way, if an attempt is made to make the steam pressure coincide with the target steam pressure by changing the amount of water supplied to the evaporator, there is the problem that the steam temperature might change accordingly. That is, as shown in FIG. 3, the efficiency of the evaporator and the efficiency of the expander of a Rankine cycle system depend on the steam temperature; in order to maximize the total efficiency of the two it is necessary to control the steam temperature at an optimum steam temperature, and if the steam temperature deviates from the optimum steam temperature as a result of the amount of water supplied being changed so as to make the steam pressure coincide with the target steam pressure, there is the problem that the total efficiency of the evaporator and the expander might be degraded.

DISCLOSURE OF THE INVENTION

The present invention has been accomplished under the above-mentioned circumstances, and it is an object thereof to control with high precision the pressure of a gas-phase working medium at the inlet of an expander in a Rankine cycle system at a target pressure without changing the amount of liquid-phase working medium supplied to an evaporator.

In order to attain this object, in accordance with an aspect of the present invention, there is proposed a Rankine cycle system that includes an evaporator for heating a liquid-phase working medium with exhaust gas of an engine so as to generate a gas-phase working medium, and a displacement type expander for converting the thermal energy of the gas-phase working

medium generated in the evaporator into mechanical energy, characterized in that, in order to make the pressure of the gas-phase working medium at the inlet of the expander coincide with a target pressure, the system includes control means for calculating a feedforward value on the basis of the target pressure and the flow rate of the gas-phase working medium at the outlet of the evaporator, calculating a feedback value by multiplying a deviation of the pressure of the gas-phase working medium at the inlet of the expander from the target pressure by a feedback gain calculated on the basis of the flow rate of the gas-phase working medium, and controlling the rotational speed of the expander on the basis of the result of addition/subtraction of the feedforward value and the feedback value.

In accordance with this arrangement, since the feedforward value is calculated on the basis of the flow rate of the gas-phase working medium at the outlet of the evaporator and the target pressure of the gas-phase working medium at the inlet of the expander, the feedback value is calculated by multiplying the deviation of the pressure of the gas-phase working medium at the inlet of the expander from the target pressure by the feedback gain calculated on the basis of the flow rate of the gas-phase working medium, and the rotational speed of the expander is controlled on the basis of the result of addition/subtraction of the feedforward value and the feedback value, it is possible to compensate for gas-phase working medium flow rate-dependent differences in the characteristics of change in the pressure of the gas-phase working medium when the rotational speed of the expander changes, and make the pressure of the gas-phase working medium at the inlet of the expander coincide with the target pressure with good responsiveness and high precision without changing the amount of liquid-phase working medium supplied to the evaporator.

A controller 20 of embodiments corresponds to the control means of the present invention.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 to FIG. 12 show a first embodiment of the present invention; FIG. 1 is a block diagram of a Rankine cycle system and a control system therefor, FIG. 2 is a map for looking up a target steam pressure from a steam energy and a target steam temperature, FIG. 3 is a graph showing the relationship between optimum steam temperature and maximum total efficiency of an evaporator and an expander, FIG. 4 is a graph showing the relationship between the pressure at the inlet and the pressure at the outlet of the expander, FIG. 5A and FIG. 5B are graphs showing changes in steam pressure when the rotational speed of the expander is changed stepwise, FIG. 6A and FIG. 6B are diagrams showing convergence of the steam pressure when the feedback gain is fixed, FIG. 7A and FIG. 7B are diagrams showing convergence of the steam pressure when the feedback gain is variable, FIG. 8 is a flowchart of a steam pressure control main routine, FIG. 9 is a flowchart of a subroutine of step S3 of the main routine, FIG. 10 is a flowchart of a subroutine of step S4 of the main routine, FIG. 11 is a map for looking up a feedforward value N_{FF} for the rotational speed of the expander from a steam flow rate Q and a target steam pressure P_O, and FIG. 12 is a table for looking up a feedback gain kp from the steam flow rate Q. FIG. 13 to FIG. 16 show a second embodiment of the present invention; FIG. 13 is a block diagram of a Rankine cycle system and a control system therefor, FIG. 14 is a flowchart of a steam pressure control main routine, FIG. 15 is a flowchart of a subroutine of step S34 of the main routine, and FIG. 16 is a map for looking up a steam specific volume V from a steam pressure P and a steam temperature T. FIG. 17 to FIG. 20 show a third embodiment of the present invention; FIG. 17 is a

block diagram of a Rankine cycle system and a control system therefor, FIG. 18 is a flowchart of a steam pressure control main routine, FIG. 19 is a flowchart of a subroutine of step S53 of the main routine, and FIG. 20 is a flowchart of a subroutine of step S54 of the main routine. FIG. 21 to FIG. 25 show a fourth embodiment of the present invention; FIG. 21 is a block diagram of a Rankine cycle system and a control system therefor, FIG. 22 is a flowchart of a steam pressure control main routine, FIG. 23 is a flowchart of a subroutine of step S72 of the main routine, FIG. 24 is a flowchart of a subroutine of step S73 of the main routine, and FIG. 25 is a flowchart of a subroutine of step S74 of the main routine.

BEST MODE FOR CARRYING OUT THE INVENTION

FIG. 1 to FIG. 12 show a first embodiment of the present invention.

As shown in FIG. 1, a Rankine cycle system for recovering the thermal energy of exhaust gas of a vehicle engine 11 is formed from an evaporator 12 for heating a liquid-phase working medium (water) with the exhaust gas of the engine 11 and generating a high temperature, high pressure gas-phase working medium (steam), a displacement type expander 13 for converting the thermal energy of the high temperature, high pressure steam generated in the evaporator 12 into mechanical energy, a condenser 14 for cooling the steam discharged from the expander 13 and condensing it into water, a tank 15 for storing water discharged from the condenser 14, a water supply pump 16 for drawing up water out of the tank 15, and an injector 17 for injecting water drawn up by the water supply pump 16 into the evaporator 12, the above being arranged in a closed circuit.

A motor/generator 18 connected to the expander 13 is disposed between the engine 11 and driven wheels; the motor/generator 18 can be made to function as a motor so as to assist the output of the engine 11, and

when the vehicle is being decelerated the motor/generator 18 can be made to function as a generator so as to recover the kinetic energy of the vehicle as electrical energy. The motor/generator 18 may be connected to the expander 13 alone, and then exhibits only the function of generating electrical energy. In the present invention, the rotational speed of the expander 13 is controlled by regulating the load (amount of electric power generated) of the motor/generator 18 so as to regulate the load imposed on the expander 13 by the motor/generator 18. Input into a controller 20 are a signal from a steam flow rate sensor 21 for detecting a steam flow rate at the outlet of the evaporator 12, and a signal from a steam pressure sensor 22 for detecting a steam pressure at the inlet of the expander 13.

The controller 20 includes target steam pressure setting means 23 for setting a target steam pressure, which is a target value for the steam pressure at the inlet of the expander 13. As shown in FIG. 2, the target steam pressure setting means 23 looks up the target steam pressure on the basis of a target steam temperature and a steam energy (steam flow rate). The steam temperature at the outlet of the evaporator 12 is controlled by regulating the amount of water supplied from the injector 17 or the water supply pump 16 to the evaporator 12 so that the steam temperature coincides with the temperature at which the total efficiency of the evaporator 12 and the expander 13 becomes a maximum (that is, an optimum steam temperature). That is, as shown in FIG. 3, the efficiency of the evaporator 12 and the efficiency of the expander 13 change depending on the steam temperature; when the steam temperature increases, the efficiency of the evaporator 12 decreases and the efficiency of the expander 13 increases, whereas when the steam temperature decreases, the efficiency of the evaporator 12 increases and the efficiency of the expander 13 decreases. There is therefore an

optimum steam temperature at which the total efficiency, which is the result of addition of the two, becomes a maximum, and the steam temperature at the outlet of the evaporator 12 is controlled so as to be at the optimum steam temperature.

The reason why the steam pressure at the inlet of the expander 13 is controlled at the target steam pressure is as follows. That is, as shown in FIG. 4, if the steam pressure at the inlet of the expander 13 coincides with the target steam pressure, the steam pressure at the outlet of the expander 13 is a pressure that is commensurate with an expansion ratio of the expander 13, but if the inlet steam pressure is too high, since the steam discharged from the outlet of the expander 13 has surplus energy remaining, there is the problem that the energy is wastefully discarded. On the other hand, if the inlet steam pressure is too low, the pressure of the steam discharged from the outlet of the expander 13 becomes negative, and there is the problem that the expander 13 carries out negative work, thus degrading the efficiency.

In order to control the steam pressure at the inlet of the expander 13 so that it is at the target steam pressure while maintaining the steam temperature at the outlet of the evaporator 12 at the optimum steam temperature, that is, without changing the amount of water supplied to the evaporator 12, the load imposed on the expander 13 by the motor/generator 18 may be regulated so as to control the rotational speed of the expander 13. As shown in FIG. 5A and FIG. 5B, when the rotational speed of the expander 13 is decreased, the steam pressure increases, whereas when the rotational speed of the expander 13 is increased, the steam pressure decreases. However, the responsiveness with which the steam pressure changes depends on the steam flow rate; when the steam flow rate is low, the responsiveness is low, and at least 100 seconds is needed for the steam pressure to reach a steady state, whereas when the

steam flow rate is high, the responsiveness is high, and it takes no more than 10 seconds for the steam pressure to reach the steady state.

If a Ti value is controlled so as to coincide with a target amount of water supplied by detecting a difference in pressure before and after the injector 17, or if the rotational speed of the water supply pump 16 is controlled by detecting a discharge pressure from the water supply pump 16, even when the rotational speed of the expander 13 changes, it is possible to maintain the amount of water supplied to the evaporator 12 constant, thereby enabling the steam temperature at the outlet of the evaporator 11 to be maintained at the optimum steam temperature.

When the steam pressure is feedback-controlled at a target steam pressure, as shown in FIG. 6A it is assumed that a feedback gain kp (proportional term) is constant; as shown in FIG. 6B, if the feedback gain kp is set so that an appropriate responsiveness can be obtained when the steam flow rate is high, sufficient responsiveness cannot be obtained when the steam flow rate is low. In contrast, as shown in FIG. 7A, by using a feedback gain kp obtained by looking it up in a gain table in which the steam flow rate is a parameter, as shown in FIG. 7B, an appropriate responsiveness can be obtained regardless of whether the steam flow rate is high or low.

That is, the gist of the present invention is that, when the rotational speed of the expander 13 is feedback-controlled so that the steam pressure at the inlet of the expander 13 coincides with a target steam pressure, the feedback gain is changed according to the steam flow rate. Specific details thereof are explained below with reference to the block diagram of FIG. 1 and the flowcharts of FIG. 8 to FIG. 10.

Firstly, in step S1 of the flowchart of FIG. 8 the steam flow rate sensor 21 detects a steam flow rate Q at the outlet of the evaporator 12, in step S2

the steam pressure sensor 22 detects a steam pressure P at the inlet of the expander 13, and in step S3 a feedforward value N_{FF} for the rotational speed of the expander 13 is then calculated. That is, in step S11 of the flowchart of FIG. 9 the feedforward value N_{FF} for the rotational speed of the expander 13 is looked up from the map of FIG. 11 using as parameters the steam flow rate Q and the target steam pressure P_O. As is clear from FIG. 11, the lower the steam flow rate Q and the greater the target steam pressure P_O, the smaller the feedforward value N_{FF}, and the higher the steam flow rate Q and the smaller the target steam pressure P_O, the larger the feedforward value N_{FF}.

Returning to the flowchart of FIG. 8, in step S4 a feedback value N_{FB} for the rotational speed of the expander 13 is calculated. That is, in step S21 of the flowchart of FIG. 10 a deviation $\Delta P = |P - P_O|$ of the steam pressure P at the inlet of the expander 13 detected by the steam pressure sensor 22 from the target steam pressure P_O set by the target steam pressure setting means 23 is calculated, and in the subsequent step S22 the gain kp is looked up by applying the steam flow rate Q detected by the steam flow rate sensor 21 to the table of FIG. 12. As is clear from the table of FIG. 12, the gain kp decreases as the steam flow rate Q increases. In step S23 the gain kp is then multiplied by the deviation ΔP, thus calculating the feedback value N_{FB} for the rotational speed of the expander 13.

Returning to the flowchart of FIG. 8, if in step S5 the steam pressure P is equal to or greater than the target steam pressure P_O, then in step S6 the feedback value N_{FB} is added to the feedforward value N_{FF} for the rotational speed of the expander 13, thus calculating a rotational speed command value N for the expander 13, and if in step S5 the steam pressure P is less than the target steam pressure P_O, then in step S7 the feedback value N_{FB} is subtracted from the feedforward value N_{FF} for the rotational speed of the

expander 13, thus calculating the rotational speed command value N for the expander 13. In this way, by controlling on the basis of the rotational speed command value N the rotational speed of the motor/generator 18, that is, the rotational speed of the expander 13, it is possible to make the steam pressure P at the inlet of the expander 13 converge to the target steam pressure P_O with good responsiveness and high precision, thereby solving the problems of the steam discharged from the outlet of the expander 13 having surplus energy remaining, and the pressure of the steam discharged from the outlet of the expander 13 becoming negative and the expander 13 carrying out negative work, thus degrading the efficiency.

FIG. 13 to FIG. 16 show a second embodiment of the present invention.

As shown in FIG. 13, the second embodiment does not include the steam flow rate sensor 21 of the first embodiment (see FIG. 1), but instead includes a water supply amount sensor 24 on the inlet side of an evaporator 12, and a steam temperature sensor 25 on the inlet side of an expander 13. Whereas in the first embodiment the steam flow rate Q is directly detected by the steam flow rate sensor 21, in the second embodiment a steam flow rate Q is calculated from a steam pressure P detected by a steam pressure sensor 22, a water supply mass flow rate Gw detected by the water supply amount sensor 24, and a steam temperature T detected by the steam temperature sensor 25, and the other arrangements and operations are the same as those of the first embodiment.

The operation of the second embodiment is explained with reference to flowcharts; firstly, in step S31 of the flowchart of FIG. 14 the steam temperature sensor 25 detects the steam temperature T at the inlet of the expander 13, in step S32 the steam pressure sensor 22 detects the steam pressure P at the inlet of the expander 13, and in step S33 the water supply

amount sensor 24 detects the water supply mass flow rate Gw to the evaporator 12.

In the subsequent step S34, the steam flow rate Q to the expander 13 is calculated without using the steam flow rate sensor 21. That is, in step S41 of the flowchart of FIG. 15 a steam specific volume V is looked up in the map of FIG. 16 using the steam temperature T and the steam pressure P as parameters. As is clear from FIG. 16, the smaller the steam pressure P and the higher the steam temperature T, the greater the steam specific volume V. In the subsequent step S42 the steam flow rate Q is calculated by multiplying the specific volume V by the water supply mass flow rate Gw detected by the water supply amount sensor 24.

When the steam flow rate Q is calculated as above, the procedure moves to steps S35 to S39 of the flowchart of FIG. 14. Since these steps are exactly the same as steps S3 to S7 of the flowchart of FIG. 8 (the first embodiment), explanation thereof is omitted so as to avoid duplication. In this way, in accordance with this second embodiment, it is possible to eliminate the steam flow rate sensor 21.

FIG. 17 to FIG. 20 show a third embodiment of the present invention.

As shown in FIG. 17, the third embodiment does not include the water supply amount sensor 24 of the second embodiment (see FIG. 13), but instead a controller 20 is equipped with a temperature control section 26. Whereas in the second embodiment the water supply amount sensor 24 detects the water supply mass flow rate Gw, in the third embodiment a steam mass flow rate Gs, which corresponds to the water supply mass flow rate Gw, is calculated from a water supply mass flow rate command G_O output by the temperature control section 26, and the other arrangements and operations are the same as those of the second embodiment.

The operation of the third embodiment is explained with reference to the flowchart; firstly, in step S51 of the flowchart of FIG. 18 a steam temperature sensor 25 detects a steam temperature T at the inlet of an expander 13, in step S52 a steam pressure sensor 22 detects a steam pressure P at the inlet of the expander 13 and, furthermore, in step S53 a steam mass flow rate Gs is calculated.

That is, in step S61 of the flowchart of FIG. 19 the water supply mass flow rate command G_O output by the temperature control section 26 for controlling the steam temperature T by controlling the amount of water supplied by an injector 17 or a water supply pump 16 is read in, and in step S62 the water supply mass flow rate command G_O is subjected to delay filter processing so as to calculate the steam mass flow rate Gs. This delay filter processing is for compensating for a time delay from the output of the water supply mass flow rate command G_O by the temperature control section 26 to the actual generation of steam by the evaporator 12.

In the subsequent step S54 of the flowchart of FIG. 18, a steam flow rate Q is calculated. A subroutine of this step S54 is shown in FIG. 20; the flowchart of FIG. 20 is substantially the same as the flowchart of FIG. 15 of the second embodiment, and the water supply mass flow rate Gw of the second embodiment is replaced by the substantially identical steam mass flow rate Gs.

When the steam flow rate Q is calculated as above, the procedure moves to steps S55 to S59 of the flowchart of FIG. 18. Since these steps are exactly the same as steps S3 to S7 of the flowchart of FIG. 8 (the first embodiment), explanation thereof is omitted so as to avoid duplication. In this way, in accordance with this third embodiment, it is possible to eliminate the water supply amount sensor 24.

FIG. 21 to FIG. 25 show a fourth embodiment of the present invention.

As shown in FIG. 21, the fourth embodiment does not include the steam temperature sensor 25 of the third embodiment (see FIG. 13), but instead a temperature control section 26 of a controller 20 outputs a steam temperature command T_O in addition to a water supply mass flow rate command G_O. A target steam pressure P_O and a steam temperature T obtained by subjecting the steam temperature command T_O to delay processing using a delay filter 2 are input into a specific volume map. A steam specific volume V looked up therein is multiplied by a steam mass flow rate Gs to calculate a steam flow rate Q. Furthermore, instead of the map of the first to the third embodiments for looking up the feedforward value N_{FF} for the rotational speed of the expander 13 using the steam flow rate Q and the target steam pressure P_O as parameters, a table for looking up a feedforward value N_{FF} for the rotational speed of an expander 13 using the steam flow rate Q alone as a parameter is provided, and the other arrangements and operations are the same as those of the third embodiment.

The steam specific volume V is shown by replacing the 'steam pressure P' of the abscissa in FIG. 16 with the 'target steam pressure P_O'.

The operation of the fourth embodiment is explained with reference to flowcharts; firstly, in step S71 of the flowchart of FIG. 22 a steam pressure sensor 22 detects a steam pressure P at the inlet of the expander 13, and in step S72 the steam mass flow rate Gs is calculated. The flowchart of FIG. 23, which is a subroutine of step S72, is substantially the same as the flowchart of FIG. 19 of the third embodiment except that a time constant τ is defined as a first time constant $\tau 1$ in order to differentiate it from a second time constant $\tau 2$, which will be described later.

In the subsequent step S73 of the flowchart of FIG. 22, the steam flow rate Q is calculated. A subroutine of this step S73 is shown in FIG. 24; in step S91 of the flowchart of FIG. 24 the steam temperature command T_O output by the temperature control section 26 is subjected to delay processing using the delay filter 2 so as to calculate the steam temperature T, and in step S92 the steam temperature T and the target steam pressure P_O output by target steam pressure setting means 23 are applied to the specific volume map so as to look up the steam specific volume V. In step S93 the steam mass flow rate Gs output by a delay filter 1 is multiplied by the steam specific volume V so as to calculate the steam flow rate Q.

Subsequently, in step S74 of the flowchart of FIG. 22, that is, in step S101 of the flowchart of FIG. 25, the steam flow rate Q is applied to an expander rotational speed table so as to look up a feedforward value N_{FF} for the rotational speed of the expander 13. Unlike the first to the third embodiments this expander rotational speed table does not use the target steam pressure P_O as a parameter, but during the process of calculating the steam flow rate Q the target steam pressure P_O is applied to the specific volume map, and as a result the target steam pressure P_O is taken into consideration. In this way, the calculated feedforward value N_{FF} for the rotational speed of the expander 13 looked up using the steam flow rate Q is proportional to the steam flow rate Q regardless of the steam temperature and the steam pressure; in practice it might not be precisely proportional to the steam flow rate Q due to the influence of steam leakage, etc., and such an error is compensated for by feedback control of the rotational speed of the expander 13.

Since the last steps S75 to S78 of the flowchart of FIG. 22 are exactly the same as steps S4 to S7 of the flowchart of FIG. 8 (the first embodiment),

explanation thereof is omitted so as to avoid duplication. In this way, in accordance with this fourth embodiment, it is possible to eliminate the steam temperature sensor 25.

Although embodiments of the present invention are explained in detail above, the present invention can be modified in a variety of ways without departing from the spirit and scope thereof.

For example, the working medium is not limited to water (steam), and another appropriate working medium may be employed.